PATENT SPECIFICATION

DRAWINGS ATTACHED

870,781

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No. 32665/59.

Application made in United States of America on September 26, 1958.

(*Patent of Addition to No.* 839,292 *dated August* 5, 1958).

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Index at Acceptance: Classes 80(2), F1, U5(B2:C1:C2:C3); **12(1),** A7D5; **79(2),** C12; **and 108(3),** S6(E1:F).

International Classification: F06c,d. B62d. F06f.

Improved rotary power transmission shaft.

COMPLETE SPECIFICATION

We, GENERAL MOTORS CORPORATION, a Company incorporated under the laws of the State of Delaware in the United States of America, of Grand Boulevard in the City of Detroit, State of Michigan, in the United States of America (Assignees of HUGH WILLIAM LARSEN) do hereby declare the invention for which we pray that a patent may be granted to us and the method by which it is to be performed, to be particularly described in and by the following statement:—

This invention relates to an improved rotary power transmission shaft, and is an improvement in the invention of our Specification No. 25,026/58. (Serial No. 839292).

According to the parent specification, a rotary power transmission device comprises a shaft which is rectilinear in unstressed condition and a pair of spaced supports for the ends of the shaft, the axes of the supports being out of alignment in at least one plane whereby to apply longitudinal bending moments to the shaft and hold the shaft in a stressed non-rectilinear position. The bending of the shaft controls first mode frequency vibration of the shaft.

The bending of the shaft does not control second mode frequency vibration, because this second mode vibration can take place without the shaft changing in length. According to the present invention, vibration absorbers for the shaft control bending vibrations in the shaft at its second mode critical speed or a sub-harmonic of this speed.

The scope of the monoply is indicated by the appended claims; how it can be carried into effect is hereinafter particularly described with reference to the accompanying drawings, in which:—

Figure 1 is a diagrammatic view of a motor vehicle transmission employing vibration absorbers according to the invention;

Figure 2 is a diagrammatic view showing first mode vibration in a curved transmission shaft;

Figure 3 is a diagrammatic view showing second mode vibration in a curved transmission shaft;

Figure 4 is a schematic view of a shaft with one type of vibration absorber;

Figure 5 is an end view of the shaft and vibration absorber shown in Figure 4;

Figure 6 is another type of vibration absorber;

Figure 7 is an end view of the shaft and vibration absorber shown in Figure 6;

Figure 8 is still another form of vibration absorber;

Figure 9 is an end view of Figure 8;

Figure 10 is still another form of vibration absorber;

Figure 11 is an end view of Figure 10;

Figure 12 is still another form of vibration absorber;

Figure 13 is an end view of Figure 12;

Figure 14 is still another form of vibration absorber.

Figure 15 is an end view of Figure 14;

Figure 16 is a cross-section of an actual vibration absorber of the type shown schematically in Figures 6 and 7;

Figure 17 is a cross-section on the lines 17-17 of Figure 16;

Figure 18 is a cross-section of an actual vibration absorber of the type shown schematically in Figures 8 and 9;

Figure 19 is a cross-section of another actual vibration absorber of the type shown schematically in Figures 8 and 9;

Figure 20 is a cross-section of an actual vibration absorber of the type shown schematically in Figures 12 and 13;

Figure 21 is a cross-ection on the line 21-21 of Figure 20; and

Figure 22 is a cross-section of another actual vibration absorber of the type shown schematically in Figures 12 and 13.

Figure 1 shows an inclined engine E connected by a curved torsion bar transmission

shaft 1 to the differential meçhanism of an inclined rear axle R, which may or may not include a speed change transmission. The forward end of the shaft 1 is connected to the engine crankshaft 3 by means of flange member 5; the rear end of the shaft 1 is connected by means of a spline connection to transmit torque to an input shaft 7 of the rear axle R.

The shaft 1 is straight prior to installation in the vehicle, but when installed is held in a stressed curve between a rear engine bearing 9 and a bearing 11 in the rear axle. With the shaft 1 held by the bearings in a circular arc, bending moment will be uniform throughout the length of the shaft. A pair of vibration absorbers 13 surrounds the shaft at specific points along its length. The type of vibration absorbers utilised determines whether these vibration absorbers 13 contact other parts of the motor vehicle, such as the body or frame.

Figure 2 shows a transmission shaft 1 held in a curved position by shaft support bearings 10 and 12. The dash lines 1A and 1B indicate extreme positions of the shaft 1 when it is vibrating at its first mode critical speed, this being the lowest or first mode natural bending frequency of the shaft. With this mode of vibration the maximum amplitude of vibration occurs at the centre point 14 of the shaft: this form of vibration is the type of vibration usually associated with long rotating power shafts, and is normally controlled by steady rest bearings at the centre of the shaft.

When the shaft is held in a stressed curved shape, as shown in the Figures, the natural frequency of vibration for the first mode is higher than that of a corresponding straight shaft. The difference between the frequencies of the shaft in its straight and curved conditions become greater as the shaft is curved more and more. beyond a certain curvature the first mode natural bending frequency of the shaft in the plane of the curvature is greater than the second mode natural frequency illustrated in Figure 3. In conventional installations utilizing straight shafts, the second mode frequency does not usually present any problem, since the second mode frequency of a straight shaft is about three times that of the first mode frequency.

The dash lines 1C and 1D in Figure 3 indicate extreme positions of the shaft when it is vibrating at the second mode frequency. It has been found that curving the shaft does not appreciably change the frequency of second mode, and, therefore, the second mode vibration shown in Figure 3 may occur at speeds of rotation of the shaft less than those causing the first mode vibration shown in Figure 2. In the second mode vibration shown in Figure 3 there is a node at the centre point 14 of the shaft, so that any steady bearing placed at the centre of the shaft would be ineffective in controlling second mode vibration.

Figures 4 to 15 show six different types of vibration absorbers for controlling the second mode of vibration: the first three are seismic vibration absorbers whereas the last three are of the grounded type. The shafts 1 are shown rectilinear, but in use they are, of course held in stressed non-rectilinear condition by the supports 10, 12.

Figures 4 and 5 show a pair of vibration absorbers 13 each including a seismic mass 15 surrounding the shaft 1 and connected to the shaft via a dashpot 17. The vibration absorbers 13 are preferably in the vicinity of the antinodes for the second mode frequency, these being at approximately the quarter points 16 on the shaft. At the quarter points the vibration absorbers will have the greatest influence on the second mode vibration. However, if it is desired simultaneously to control the first mode vibration, which has its maximum amplitude at the centre of the shaft, the vibration absorbers are positioned between the quarter points and the centre of the shaft. The effect of the vibration absorbers on the first mode vibrations will then be increased and the effect on the second mode vibrations will be decreased. The best position for the vibration absorbers will vary from installation to installation, and can easily be determined experimentally.

With the arrangement shown in Figures 4 and 5, movement of the shaft 1 from its normal position into either the first or second mode shapes can occur only under control of the dashpot 17, which may be a conventional dashpot or any equivalent vibration energy dissipating device, such as a viscous damper, a friction damper or a butyl or natural rubber device able to dissipate energy internally. Since the vibrational energy of the shaft 1 is dissipated in the dashpot 17, whose movement is resisted by the seismic mass 15, the amplitude of vibration of the shaft 1 will be minimised since the vibrational energy cannot build up in the shaft 1.

Figures 6 and 7 illustrate a seismic vibration absorber that utilises a combined damper and spring. A seismic mass 15 again surrounds the shaft 1 and is connected to the shaft through a dashpot or equivalent energy dissipating device 17, but there is an additional connection through a spring member 19. In practice the functions of the dashpot 17 and the spring member 19 may be shared by a single member, which may be formed of butyl or other rubber, between the seismic mass 15 and the shaft 1. In the construction shown in Figures 6 and 7, only a portion of the vibrational energy is dissipated in the dashpot 17, the remainder being stored in the spring member 19 and

returned to the shaft during the next half cycle of vibration.

In both the constructions shown in Figures 4 to 7, the seismic mass 15 is allowed to rotate with the shaft 1, with the resulting advantages of no physical connection between the shaft and the other portions of the motor vehicle or other power installation. Vibrational forces thus cannot be transmitted to the vehicle and felt by the occupants.

The rotating mass type of vibration damper shown in Figures 4 to 7 operates very well at low shaft speeds. Where a rotating mass vibration damper is used on a shaft driven at very high speeds, however, the mass itself can become unbalanced and, after overcoming the spring force of the vibration absorber, tend to fly radially outwards of the shaft. This would result in an unbalanced condition, which would be undesirable. Figures 8 and 9 illustrate a construction similar to that shown in Figures 6 and 7 except that in the embodiment of Figures 8 and 9 the seismic mass 15 is held against rotation by a stop rod 21 that engages some fixed part of the vehicle. With such a construction there must be some provision for relative rotation between the shaft 1 and the seismic mass 15, for example bearings or other antifriction elements between the shaft and the dashpot 17 and springs 19 or between the dashpot 17 and springs 19 and the seismic mass 15.

Shaft vibration can also be controlled by vibration dampers connected to a fixed element, for example a non-rotating part of the vehicle such as the body, frame or engine housing. Figures 10 and 11 show schematically such a device, with dashpots 23 connecting the shaft to the fixed element. As in Figures 4 and 5, vibrational energy is dissipated in the dashpots 23 and hence cannot build up in the shaft 1. As in the devices shown in Figures 4 to 9, the dashpot 23 may be any device in which there is internal dissipation of energy. By appropriate selection of the damping characteristics of the dashpots, the amount of vibrational force transmitted to the fixed element will be kept a minimum.

Figures 12 and 13 show another form of vibration control, with springs 25 in parallel with the dashpots 23 between the shaft 1 and a fixed element 27. Only a portion of the energy is dissipated in the dashpots 23; the remainder is stored in the springs 25.

Figures 14 and 15 illustrate another construction, which includes a pair of permanent magnets or electromagnets 29 connected to the fixed element 27. The poles of the magnets 29 are so arranged that vibration of the rotating shaft 1 out of its rest position causes the shaft to cut through the flux lines of the magnets: the resulting magnetic force created in the shaft is such as to tend to return the shaft to its rest position in which no flux lines are cut through by the rotating shaft. This construction has the advantage that there is no physical connection between the shaft and the vibration damper itself.

Figs. 16 to 22 illustrate five different actual constructions in which a seismic mass 31 controls the second and first mode vibrations in the rotating shaft. The first construction, shown in Figs. 16 and 17, is equivalent to that shown schematically in Figs. 6 and 7, and includes rubber shims 33 surrounding the shaft 1, a split metal ring 35 surrounding the shims 33 and a pair of attaching rings 37 secured to the split rings 35 by means of set screws 38. A vibration absorber support sleeve 39 is secured to the attaching ring 37 by means of set screws 40. A pair of butyl rubber rings 41 acting as a combined vibration absorber and spring is fitted between the support sleeve 39 and the seismic mass 31, which is cylindrical. The rubber rings 41 have holes 43, as seen in Fig. 17; the shape and size of the rings 41 are chosen to give the desired damping and spring characteristics.

Fig. 18 illustrates an actual vibration absorbing device of the type shown schematically in Figs. 8 and 9, and includes a support sleeve 45 mounted on the shaft 1 by means of rubber shims 33, split rings 35 and set screws 38 similar to the manner in which the attaching rings 37 are mounted on the shaft 1 in the vibration absorber of Figs. 16 and 17. A seismic mass 59 surrounds the whole assembly and is connected to the support sleeve 45 by means of a ball bearing 51 which includes an inner race 53, an outer race 55 and balls 57. A pair of rubber rings 47 formed of butyl rubber or equivalent material surrounds the supporting sleeve 45; these rings are themselves surrounded and held in place by a pair of attaching rings 58 having flanged portions 60 to receive the inner race 53 of the ball bearing 51. A stop rod 61 prevents rotation of the seismic mass 59.

Figure 19 shows another construction of the type shown schematically in Figures 8 and 9, relative rotation between the shaft 1 and the seismic mass being taken up at an antifriction bearing between the shaft and the vibration absorber rather than between the vibration absorber and the mass as in the previous embodiment shown in Figure 18. In the Figure 19 construction a rubber sleeve 63 is fitted on the shaft 1, the sleeve 63 being surrounded by a metal retainer sleeve 64 carrying a ball bearing 51. A pair of annular stepped support rings 65 surrounds the ball bearing 51, and these rings are secured together by means of bolts 66. These support rings 65 have external

flanges 67 with grooves 68 to receive vibration absorbing rings 69 formed of butyl rubber, for example. The rings 69 carry a seismic mass 71 which includes a pair of annular end plates 72 and a cylindrical member 74. The stop rod 61 attached to the seismic mass 71 can engage a fixed part of the vehicle to prevent rotation of the seismic mass 71. As before, the damper rings 69 are designed to provide the desired vibration damping and spring effect.

Figures 20 and 21 illustrate an actual construction of a vibration absorber of the type shown schematically in Figures 12 and 13. A shim 73 surrounds the shaft 1. The shim 73 may be pressed on the shaft, or glued or otherwise secured to it. A ball bearing 51 surrounds the shim 73 and is itself surrounded by a pair of retainer rings 75. A rubber damper 79 forced over the retainer rings 75 is enclosed by a combined U-shaped support and retainer 77. The damper member 79 is designed to have the proper damping and spring characteristics.

Another actual construction of the type shown schematically in Figures 12 and 13 is shown in Figure 22. In this construction, a shim 73 is secured to the shaft 1 and is itself surrounded by a metal ring 81 forming the inner race for a needle roller bearing 83. The outer race of the needle roller bearing 83 is surrounded by a folded rubber ring 85 forming an annular chamber 87 filled with silicone fluid or other viscous medium. An outer ring 89 seals the ring 85 and is secured to a fixed part of the vehicle to support the ring 85. The silicone fluid provides damping additional to that provided by the rubber, vibrational energy being dissipated within the fluid as well as within the rubber.

WHAT WE CLAIM IS:—

1. A rotary power transmission device comprising a shaft which is rectilinear in unstressed condition and a pair of spaced supports for the ends of the shaft, the axes of the supports being out of alignment in at least one plane whereby to apply longitudinal bending moments to the shaft and hold the shaft in a stressed non-rectilinear position, and vibration absorbers for the shaft to control bending vibrations in the shaft at its second mode critical speed or a sub-harmonic of this speed.

2. A rotary power transmission device comprising a shaft which is rectilinear in unstressed condition and a pair of spaced supports for the ends of the shaft, the axes of the supports being out of alignment in at least one plane whereby to apply longitudinal bending moments to the shaft and hold the shaft in a stressed non-rectilinear position, and a plurality of vibration absorbers connected to the shaft at specific points along its length to control bending vibrations in the shaft at its second mode critical speed or a sub-harmonic of this speed.

3. A power transmission device according to claim 2, wherein the vibration absorbers comprise seismic masses connected to the shaft via vibrational energy dissipating means.

4. A power transmission device according to claim 2, wherein the vibration absorbers comprise seismic masses connected to the shaft via vibrational energy dissipating means and via spring means with a specific spring rate.

5. A power transmission device according to claim 3 or 4, including means for preventing rotation of the seismic masses with the shaft.

6. A power transmission device according to claim 2, wherein the vibration absorbers comprise vibrational energy dissipating means connecting the shaft to a member which is fixed with respect to the shaft supports.

7. A power transmission device according to claim 2, mounted in a motor vehicle with the ends of the shaft connected respectively to the crankshaft of the vehicle motor and to a drive gear for the rear wheels.

8. A power transmission device according to claim 7, wherein the vibrational absorbers comprise bearings spaced along the shaft and a vibration damper connecting each bearing to a stationary part of the vehicle.

9. A power transmission device according to claim 8, wherein each vibration damper includes a natural or synthetic rubber member capable of both storing and dissipating vibrational energy from the shaft.

10. A power transmission device according to claim 9, wherein each rubber member comprises butyl rubber in the form of an annular chamber which surrounds the bearing and is filled with a viscous fluid.

11. A power transmission device according to any one of claims 2 to 10, wherein the vibration absorbers are spaced along the shaft to control the shaft at its first mode critical speed also.

12. A power transmission device according to claim 11, wherein the vibration absorbers are disposed at a distance of between one-quarter and one-third of the shaft length from each end of the shaft.

13. A rotary power transmission device substantially as hereinbefore particularly described and as shown in Figures 4 and 5 of the acompanying drawings.

14. A rotary power transmission device substantially as hereinbefore particularly described and as shown in Figures 6 and 7 of the accompanying drawings.

15. A rotary power transmission device

substantially as hereinbefore particularly described and as shown in Figures 8 and 9 of the accompanying drawings.

16. A rotary power transmission device substantially as hereinbefore particularly described and as shown in Figures 10 and 11 of the accompanying drawings.

17. A rotary power transmission device substantially as hereinbefore particularly described and as shown in Figures 12 and 13 of the accompanying drawings.

18. A rotary power transmission device substantially as hereinbefore particularly described and as shown in Figures 14 and 15 of the accompanying drawings.

19. A rotary power transmission device substantially as hereinbefore particularly described and as shown in Figures 16 and 17 of the accompanying drawings.

20. A rotary power transmission device substantially as hereinbefore particularly described and as shown in Figure 18 of the accompanying drawings.

21. A rotary power transmission device substantially as hereinbefore particularly described and as shown in Figure 19 of the accompanying drawings.

22. A rotary power transmission device substantially as hereinbefore particularly described and as shown in Figures 20 and 21 of the accompanying drawings.

23. A rotary power transmission device substantially as hereinbefore particularly described and as shown in Figure 22 of the accompanying drawings.

E. WILLIAMSON,
Chartered Patent Agent.

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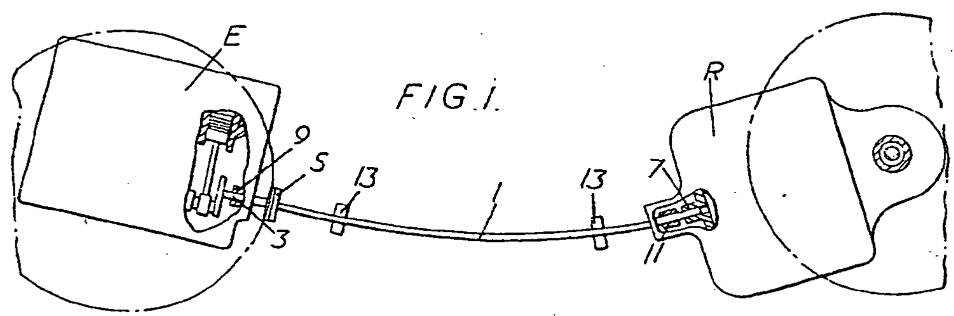

FIG.1.

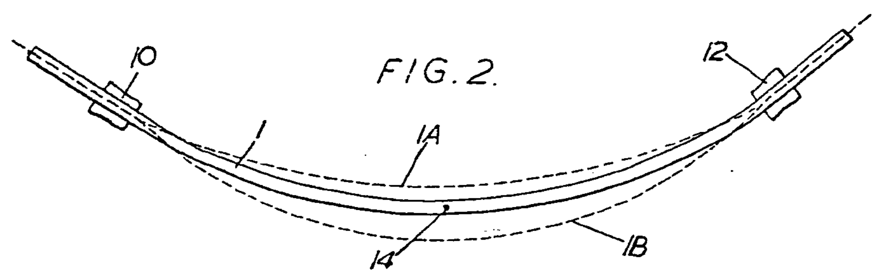

FIG.2.

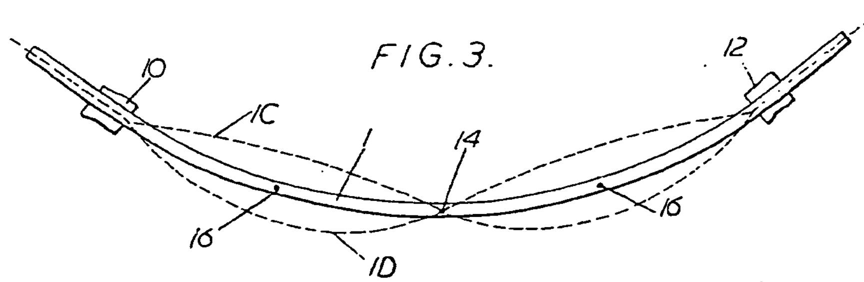

FIG.3.

870,781 COMPLETE SPECIFICATION
4 SHEETS
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SHEETS 1 & 2

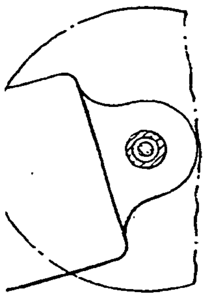


FIG.4.

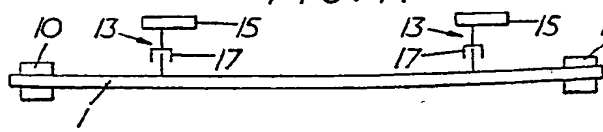


FIG.5.

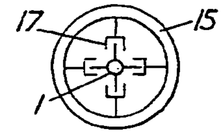


FIG.6.

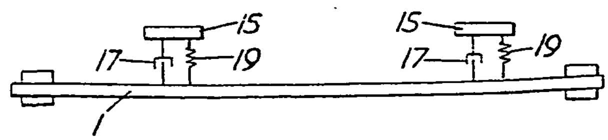


FIG.7.

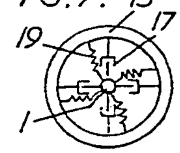


FIG.8.

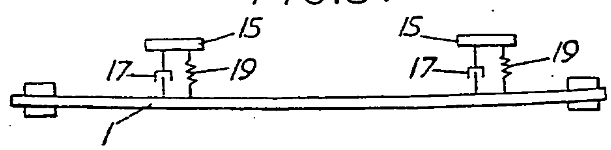
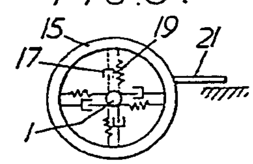


FIG.9.



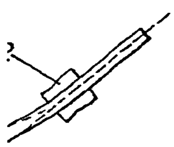


FIG.10.

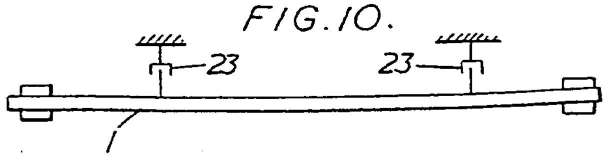


FIG.11.

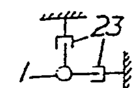


FIG.12.

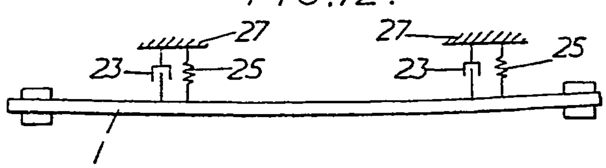
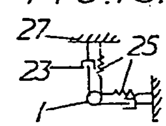


FIG.13.



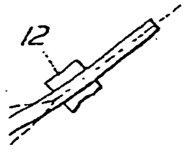


FIG.14.

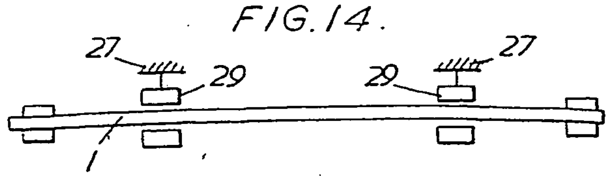
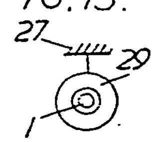


FIG.15.



870,781 4 SHEETS

COMPLETE SPECIFICATION

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SHEETS 1 & 2

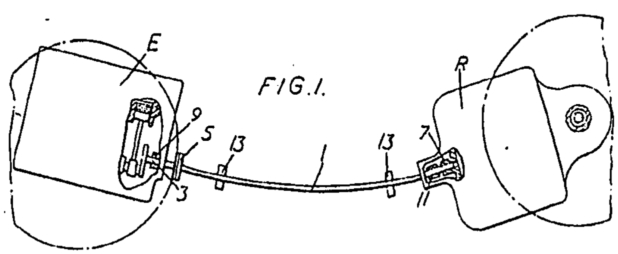


FIG.1.

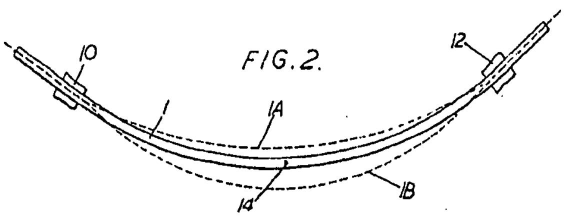


FIG.2.

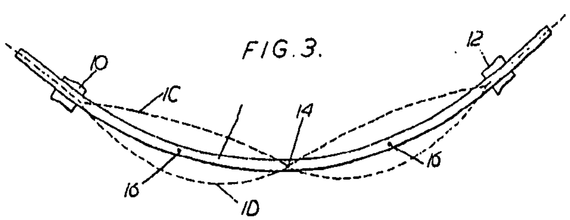
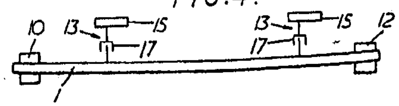


FIG.3.

FIG.4.



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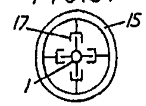

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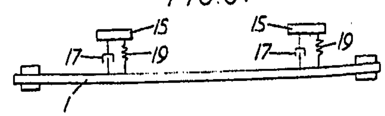

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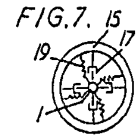

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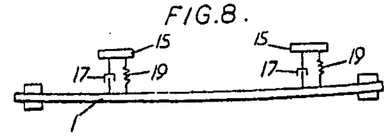

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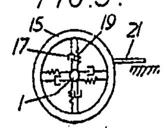


FIG.10.

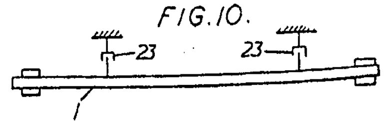
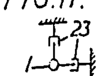


FIG.11.



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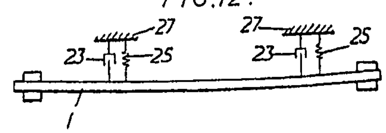

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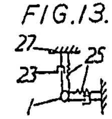

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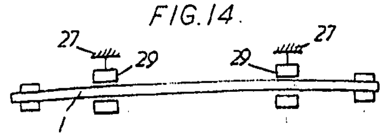

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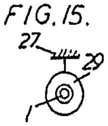
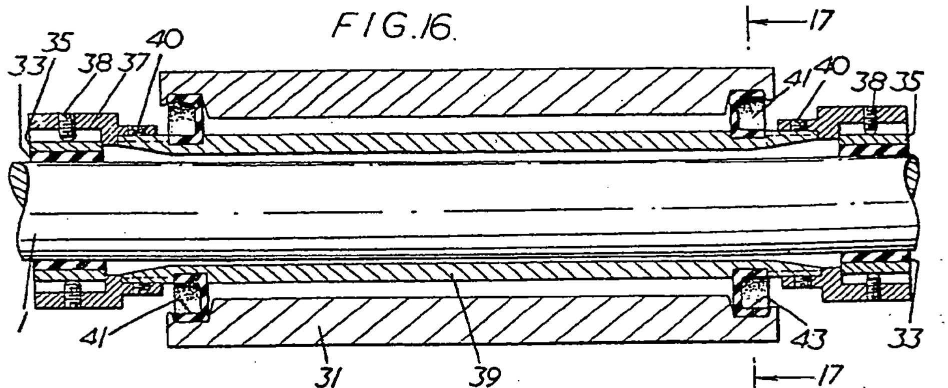


FIG. 16.



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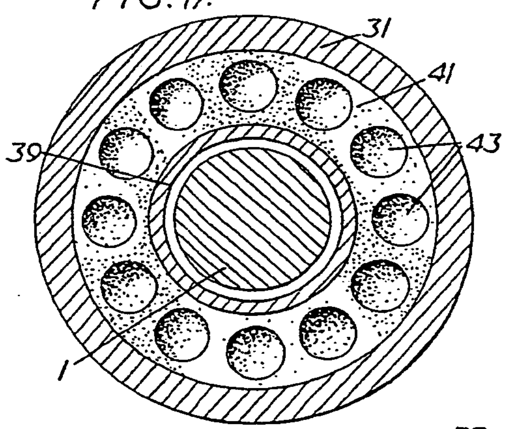
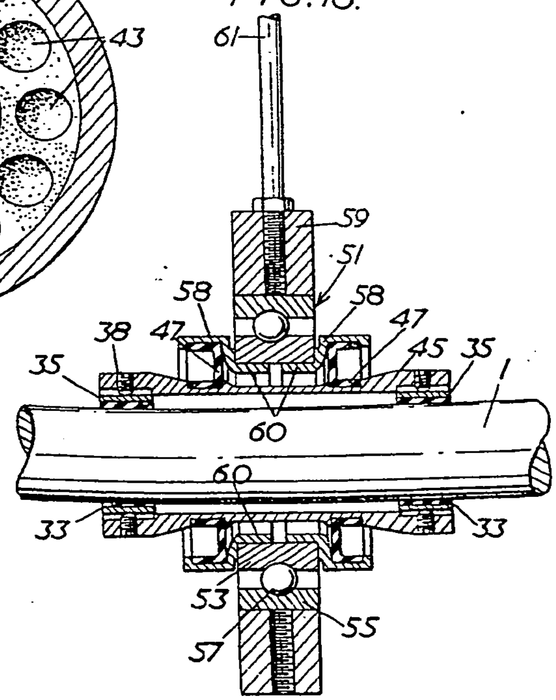


FIG. 18.



870,781 COMPLETE SPECIFICATION
4 SHEETS

This drawing is a reproduction of the Original on a reduced scale.
SHEETS 3 & 4

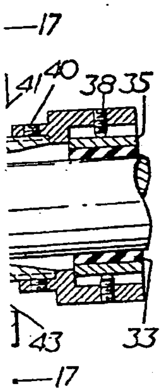


FIG. 19.

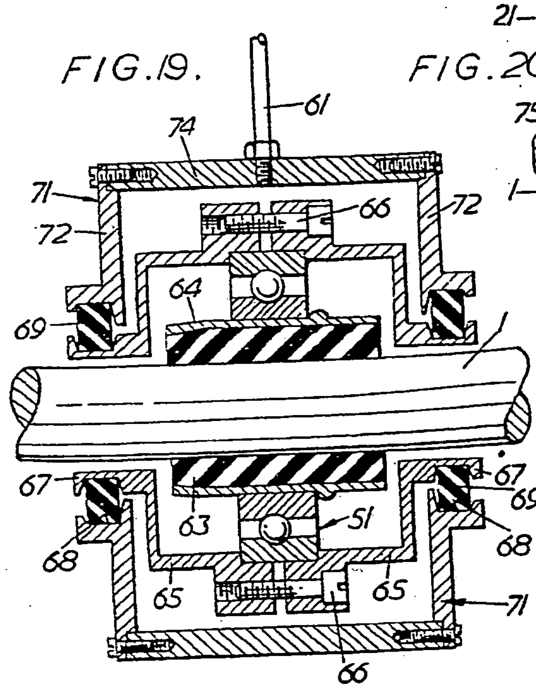
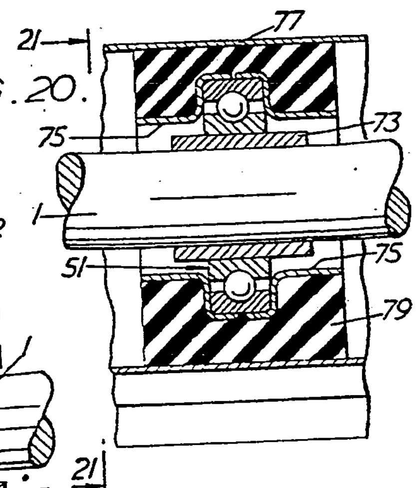


FIG. 20.



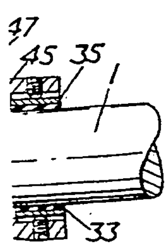


FIG. 21.

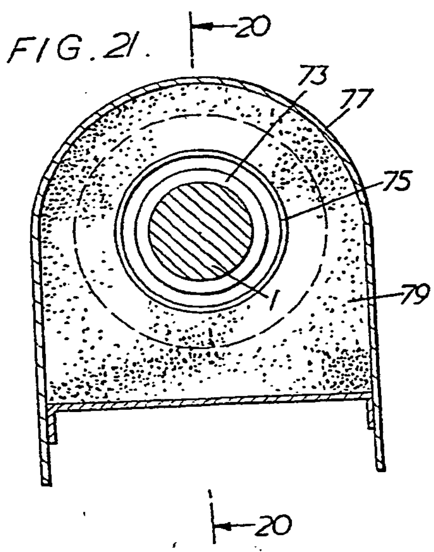
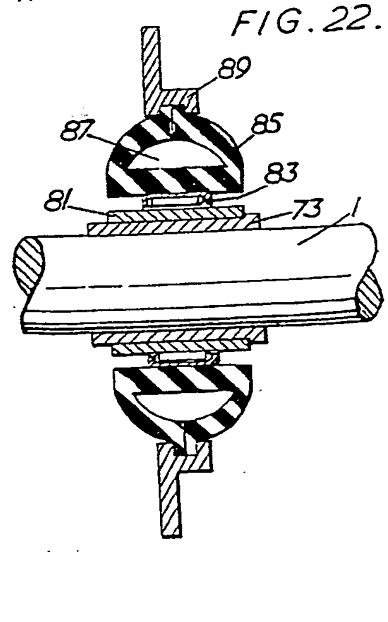


FIG. 22.



870,781 COMPLETE SPECIFICATION
4 SHEETS This drawing is a reproduction of the Original on a reduced scale.
SHEETS 3 & 4

FIG.16.

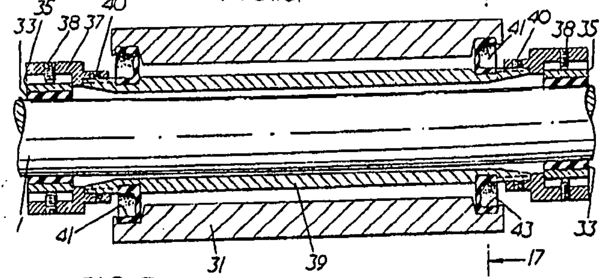


FIG.17.

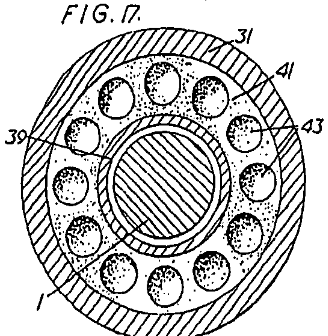


FIG.18.

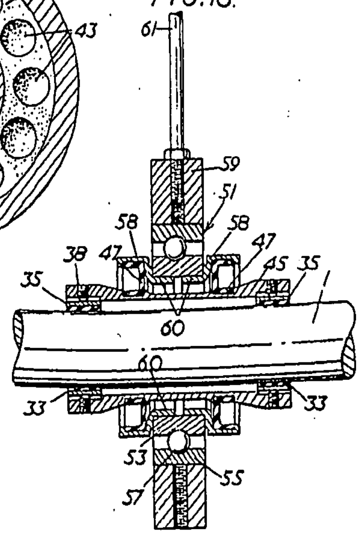


FIG.19.

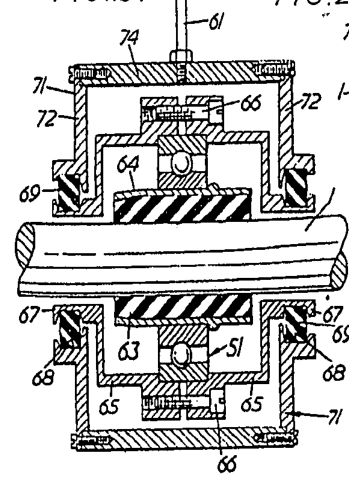


FIG.20.

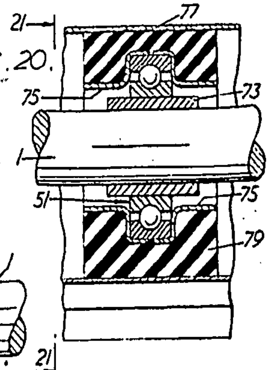


FIG.21.

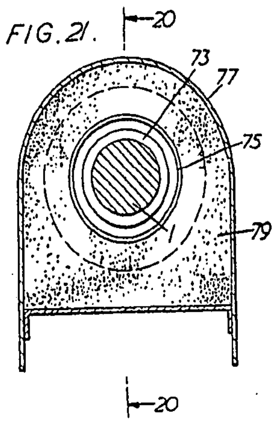


FIG.22.